

Optimal design of the regenerative gas turbine engine with isothermal heat addition

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Abstract

A regenerative gas turbine engine, with isothermal heat addition, working under the frame of a Brayton cycle has been analyzed. With the purpose of having a more efficient small-sized gas turbine engine, the optimization has been carried out numerically using the maximum power (MP) and maximum power density (MPD) method. The effects of internal irreversibilities have been considered in terms of the isentropic efficiencies of the turbine and compressor and of the regenerator efficiency. The results summarized by figures show that the regenerative gas turbine engine, with isothermal heat addition, designed according to the maximum power density condition gives the best performance and exhibits highest cycle efficiencies. © 2001 Elsevier Science Ltd. All rights reserved.

1. Introduction

The use of advanced cycles for large scale power generation has received increasing attention in recent years. These cycles have been developed to take advantage of the gas turbine's thermodynamic characteristics. Heppenstall [1] describes and compares their thermodynamic and economic characteristics in order to establish their relative importance to future power generation markets. Increasing the thermal efficiency of the power generation process is the most important aim of the new designs since the improved efficiency results in improved commercial performance leading to lower levels of pollution for a given output of electricity. Hernandez et al.

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Nomenclature

CCC	the converging combustion chamber
c	the specific heat
JB	Joule–Brayton engines
k	the ratio of specific heat capacities
M	Mach number
M_i	Mach number at the CCC inlet
M_e	Mach number at the CCC exit
MP	the maximum power
MPD	maximum power density
$\dot{m}$	the mass flow rate of the working fluid
RCC	the regular combustion chamber
r_p	the cycle pressure ratio, $r_p = P_2/P_1$
r_t	the isothermal pressure drop ratio, $r_t = P_4/P_3$
r_v	the cycle compression ratio, v_5/v_2
$r_{p,mp}$	the optimum cycle pressure ratio at the MP conditions
$r_{p,mpd}$	the optimum cycle pressure ratio at the MPD conditions
T	temperature
$\dot{W}$	the net power
$\dot{W}_d$	the power density
W^*	the dimensionless power
W_d^*	the dimensionless power density
W_{max}^*	the maximum net power output
$W_{d,max}^*$	the maximum power density
W^*/W_{max}^*	the normalized power
$W_d^*/W_{d,max}^*$	the normalized power density
α	the temperature ratio, $\alpha = T_1/T_3$
η	the thermal efficiency of the cycle
η_c	the compressor's isentropic efficiency
η_r	the regenerator's efficiency
η_t	the turbine isentropic efficiency

[2] have considered the behavior of the power output of a gas turbine power plant with multiple reheating and intercooling stages to obtain higher efficiencies and higher powers. Göktun and Yavuz [3] have shown that the application of an isothermal heat addition process to the Brayton cycle yields significant increases of the thermal efficiency.

In thermal engineering, power systems are optimized to get optimal thermodynamic operating conditions. Before the optimization process, the power system is modeled by considering the different irreversibilities to get better realistic models [4,5]. The minimization of the irreversibilities associated with the hot and cold heat exchangers and regenerator has been studied by Wu et al. [6]. Theoretical analysis of

a closed Brayton cycle with internal irreversibilities has been carried out by Chen et al. [7]. Radcenko et al. [8] have considered the open Brayton cycle and shown that the thermodynamic performance can be optimized by fine tuning the fuel flow rate and subsequent distribution of pressure drops. A set of optimum power expressions for the open irreversible Brayton and open Joule–Brayton heat engine cycles with a finite interactive heat source, has been obtained by Blank [9]. Sahin et al. [10] have introduced an optimization criterion called the maximum power density (MPD) and investigated the optimal performance conditions for reversible [10] and irreversible [11] non-regenerative, and irreversible regenerative reheating [12] Joule–Brayton (JB) engines. Medina et al. [13] applied the MPD analysis to a regenerative JB cycle. These studies have shown that the non-regenerative JB cycles have higher thermal efficiencies at MPD conditions than those at the maximum power (MP) conditions whereas the regenerative cycle efficiency is lower than that under MP conditions. The reheating has improved the performance and a smaller engine size advantage is obtained at certain values of temperature ratio and isentropic efficiency.

In this study, a regenerative gas turbine engine, with isothermal heat addition, working under the frame of the Brayton cycle has been studied with the purpose of developing more efficient small-sized gas turbine engines. The focus of this study is on the determination of optimal operating conditions and design parameters of the engine under the maximum power and the maximum power density conditions. The optimization has been carried out by using the maximum power density method and the irreversible turbine, compressor and regenerator have been considered. The results have been derived numerically and summarized by figures.

2. Theoretical model

A theoretical model for a regenerative Brayton cycle with isothermal heat addition has been derived and a detailed numerical example has been reported by Göktun and Yavuz [3]. The main characteristics of the hypothetically modified Brayton cycle, shown in Fig. 1, are summarized comprehensively for the completeness of the present study as follows:

- i. The working fluid of the cycle (air, air and combustion gases) is considered as an ideal gas with constant specific heat. The flow of the working fluid is one-dimensional steady flow.
- ii. The basic components of the cycle are the compressor, the regenerator, the regular combustion chamber (RCC), the converging combustion chamber (CCC) and the turbine. The constant pressure heat addition process takes place in the RCC. In the CCC, the isothermal heat addition takes place, which has been proposed for significant efficiency improvements in regenerative gas turbine engines.
- iii. The compression (1–2) and expansion (4–5) processes are adiabatic and irreversible. The deviations from the isentropic processes are accounted for by the isentropic efficiencies. The turbine's isentropic efficiency, η_t , gives the

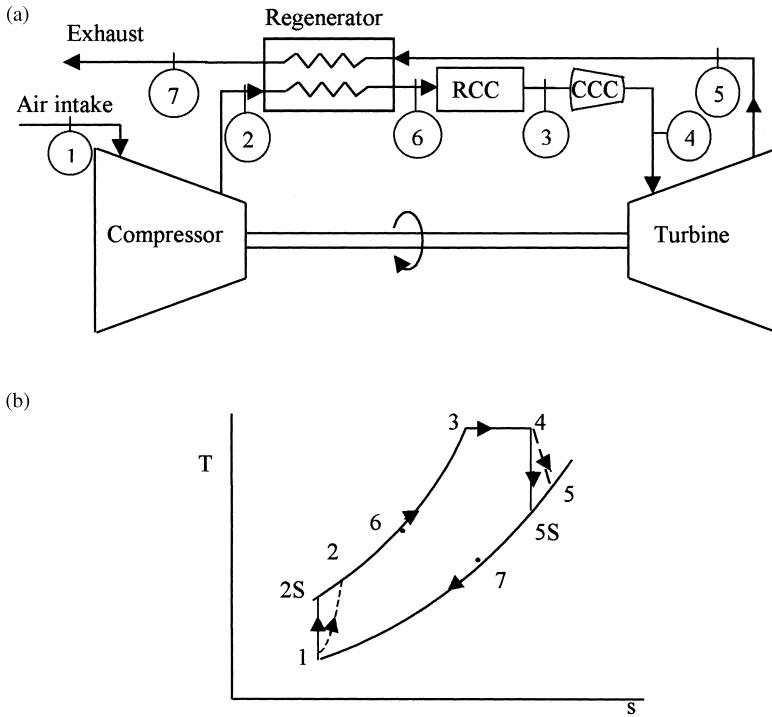


Fig. 1. (a) Schematic of equipment and (b) T-s diagram of the regenerative Brayton cycle with isothermal heat addition (1-2S, 4-5S: reversible adiabatic processes, 1-2, 4-5: irreversible processes).

ratio of the actual expansion process producing power to the isentropic one. The isentropic efficiency of the compressor, η_c , is defined as the ratio of the power requirement during the isentropic compression to that of the actual compression. η_t and η_c have the values between zero and unity.

- iv. The heat transfer in the real regenerative heat exchanger is obtained by using the regenerator efficiency, η_r defined as follows

$$\eta_r = (T_6 - T_2)/(T_5 - T_7) \quad (1)$$

which is the ratio of the actual heat transfer (2-6) to the maximum idealistic amount of heat transfer (5-7).

The value of the η_r varies from 0 to 1.

- v. There is no pressure drop in the RCC and during the steady flow of the hot stream (5-7) and cold stream (2-6) in the regenerator.
- vi. The temperature during the heat addition along the converging duct is constant, that is why the kinetic energy increases, thus it results in an increase in Mach number due to energy conservation.
- vii. The cycle has a pressure ratio, which is the ratio of the maximum pressure of the cycle obtained at the compressor exit (2) to the minimum pressure of the

cycle at the turbine exit which corresponds to the compressor inlet (1), $r_p = P_2/P_1$.

- viii. The isothermal heat addition process through the CCC (3–4) is characterized by the isothermal pressure drop ratio which gives the ratio of the pressure at the inlet to the pressure at the outlet of the CCC, $r_t = P_4/P_3$. The value of the isothermal pressure drop ratio r_t is obtained in the range 1.538–1.052.

The net power, $\dot{W}$ and the efficiency, η are written by [3] for the Brayton cycle with the isothermal heat addition under these conditions as

$$\dot{W} = \dot{m} c_p T_3 \{ [\eta_t(1 - b^{-1}) + 0.5(k - 1)M_4^2] - [\alpha(a - 1)/\eta_c] \} \quad (2)$$

$$\eta = \frac{[\eta_t(1 - b^{-1}) + 0.5(k - 1)M_4^2] - [\alpha(a - 1)/\eta_c]}{\left[(1 - \alpha) - \frac{\alpha}{\eta_c}(a - 1) \right] (1 - \eta_r) + \eta_r \eta_t \left(1 - \frac{1}{b} \right) + 0.5(k - 1)(M_e^2 - M_i^2)} \quad (3)$$

where $\dot{m}$ is the mass flow rate of the working fluid, c_p is the constant-pressure specific heat capacity, k is the ratio of specific heat capacities, $\alpha = T_1/T_3$ is the ratio of minimum temperature T_1 obtained at the compressor inlet to the maximum temperature T_3 at exit of the converging combustion chamber (CCC), M_i and M_e are the Mach numbers at the CCC inlet and exit, respectively. In Eq. (2), a and b are defined as

$$a = r_p^{(k-1)/k} \quad (4)$$

$$b = \left(\frac{r_p}{r_t} \right)^{(k-1)/k} \quad (5)$$

In the numerical analysis of maximum power density, dimensionless forms must be preferred. The dimensionless power is obtained by using Eq. (2) as

$$W^* = \frac{\dot{W}}{\dot{m} c_p T_3} \quad (6)$$

where the denominator represents the enthalpy of the working fluid having the maximum temperature. The dimensionless power density is defined as

$$W_d^* = \frac{W^*}{r_v} \quad (7)$$

where r_v is the cycle compression ratio which is the ratio of the maximum volume at the turbine exit to the minimum volume found at the compressor exit, i.e. $r_v = v_5/v_2$. The cycle compression ratio (r_v) can be connected to the cycle pressure ratio (r_p) and

the isothermal pressure ratio (r_t) via the ideal gas relations at each nodal point and properties of the processes. The relation is obtained as

$$r_v = \frac{1}{\alpha} r_p^{1/k} r_t^{(1-1/k)} \quad (8)$$

Since the main parameter effecting the thermal efficiency of the Brayton cycle is the pressure ratio, the dimensionless power density has been expressed in terms of pressure ratios rather than volumetric compression ratio by inserting Eq. (8) into Eq. (7). This form preferred in the present study enables us to find comprehensive results.

The maximization of the power of the regenerative Brayton cycle with isothermal heat addition is accomplished numerically by setting the first derivative of the power with respect to cycle pressure ratio equal to zero as described by Erbay and Yavuz [14,15]. This yields the optimized pressure ratio corresponding to pressure ratio at the maximum power output at the prescribed conditions. Maximum power density is obtained by following the same procedure.

3. Results and discussion

In this study, the maximum power and the maximum power density working conditions of the Brayton engine with isothermal heat addition were determined and compared under different conditions with the objective of having a more efficient small-sized gas turbine engine.

The results of the numerical analysis are summarized with figures by considering following:

- i. The optimum cycle pressure ratios are obtained by maximizing the power and power density for given conditions.
- ii. The optimum cycle pressure ratio found for power, $r_{p,mp}$ is inserted into Eq. (6) to obtain the maximum net power output, W_{max}^* . The maximum power density $W_{d,max}^*$ is also obtained by using the optimum cycle pressure ratio, $r_{p,mpd}$.
- iii. The power and the power density values are used in normalized forms defined by the use of maximized values for the corresponding cases as W^*/W_{max}^* and $W_d^*/W_{d,max}^*$.
- iv. In order to examine the effects of irreversibilities on the system performance obtained for maximum power and maximum power density conditions, the thermal efficiencies are obtained by considering various values for thermal efficiency of the turbine, compressor and regenerator, individually.
- v. The effects of the cycle temperature ratio, α , and the isothermal pressure ratio, r_t , on the thermal efficiency are determined.
- vi. For k , the standard value of 1.4 is taken.
- vii. The effects of the variations of the flow conditions in the converging combustion chamber are investigated by taking different values for Mach number

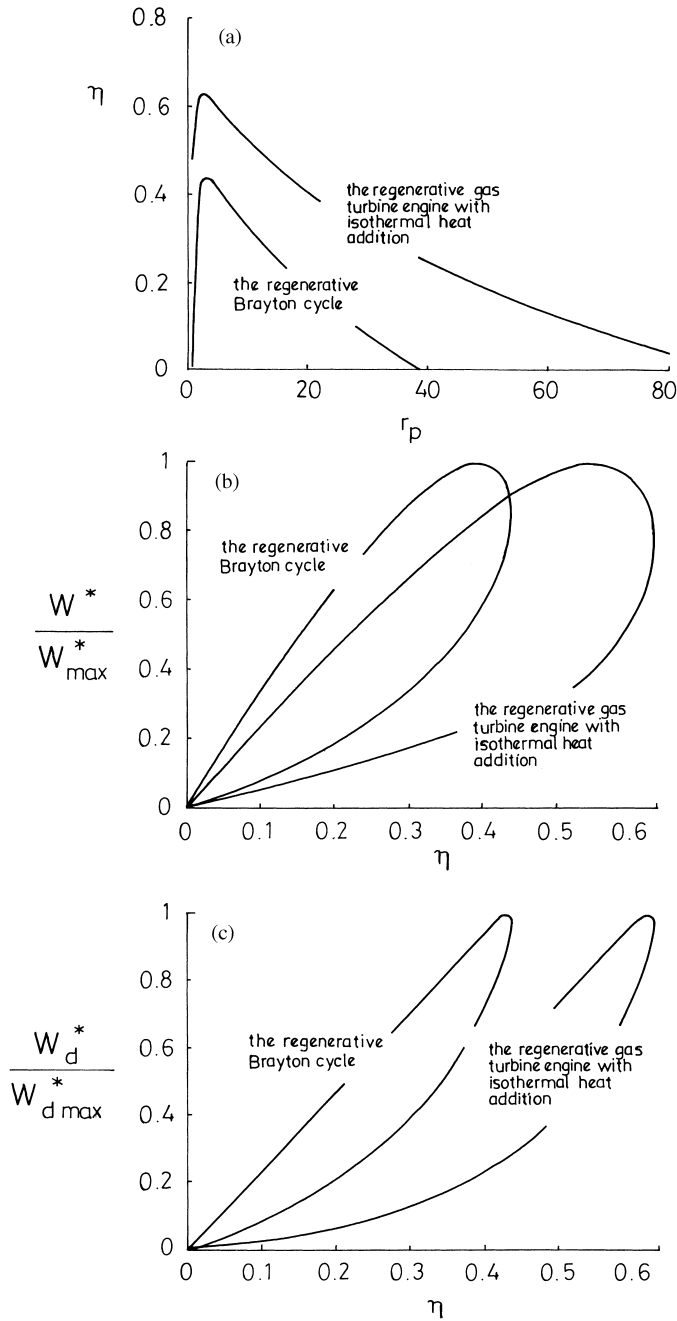


Fig. 2. The effect of heat addition on the (a) cycle efficiency with respect to the cycle pressure ratio r_p , (b) net power output W^* and (c) power density $W_{d\max}^*$, in accordance with the cycle efficiency ($\eta_t = 0.95$, $\eta_c = 0.90$, $\eta_r = 0.85$, $\alpha = 0.3$, $r_t = 1.25$).

at the exit of the CCC. Except for this analysis, the Mach numbers at the inlet and exit of the CCC are taken as 0.2 and 0.8, respectively.

Initially, the effect of a hypothetical modification obtained by an isothermal heat addition process on the maximum power and the maximum power density are investigated and summarized in Fig. 2. Fig. 2(a) yields the result of the isothermal heat addition process on the thermal efficiency with respect to cycle pressure ratio. Fig. 2(b) and (c) present the changes on the maximum power and the maximum

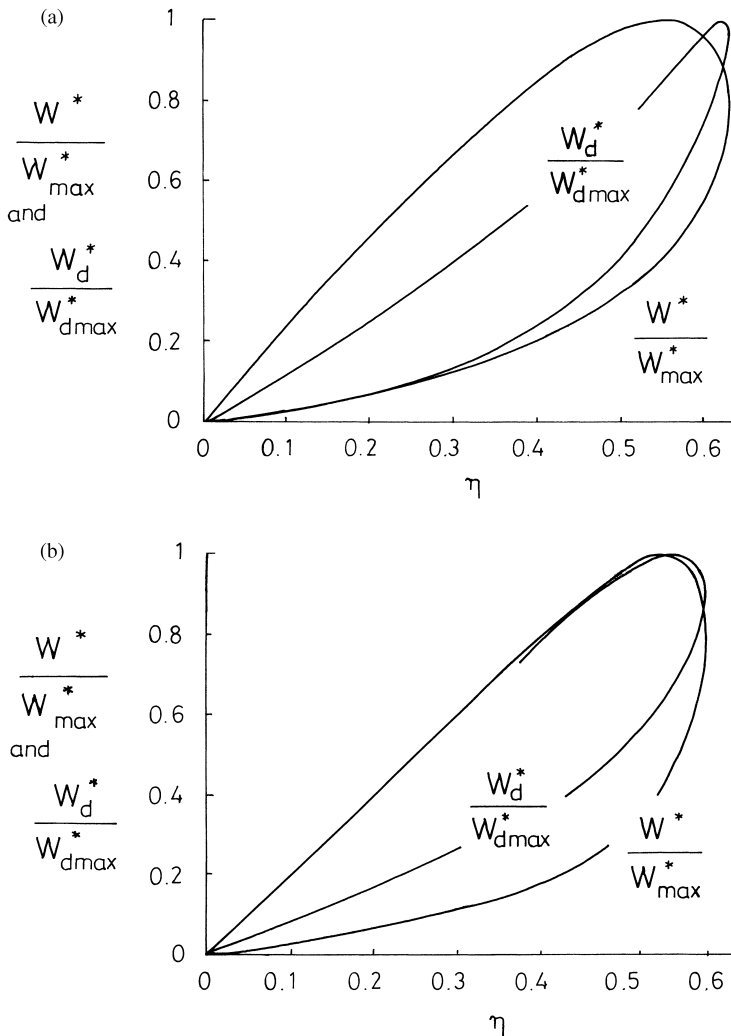


Fig. 3. The normalized power and normalized power density versus efficiency η under the conditions of (a) $\eta_t = 0.95$, $\eta_c = 0.90$, $\eta_r = 0.85$, $\alpha = 0.3$, $r_t = 1.25$ and (b) $\eta_t = \eta_c = 0.85$, $\eta_r = 0.80$, $\alpha = 0.2$, $r_t = 1.25$.

power density considering the thermal efficiency of the cycle. It is obviously seen that the modified cycle has higher efficiencies, higher power and power density levels than that of the classical regenerative Brayton cycle.

Fig. 3(a) summarizes the typical situation of the engines designed by considering the maximum power and the maximum power density conditions and shows that the engines designed by considering the maximum power density conditions have higher efficiencies. It was noted by Medina et al. [13] that the efficiency of the regenerative engine working under the maximum power density condition is smaller than the efficiency of the regenerative engine working under the maximum power condition. Fig. 3(b) compares the normalized power, $W^*/W_{\max}^*$ and normalized power density,

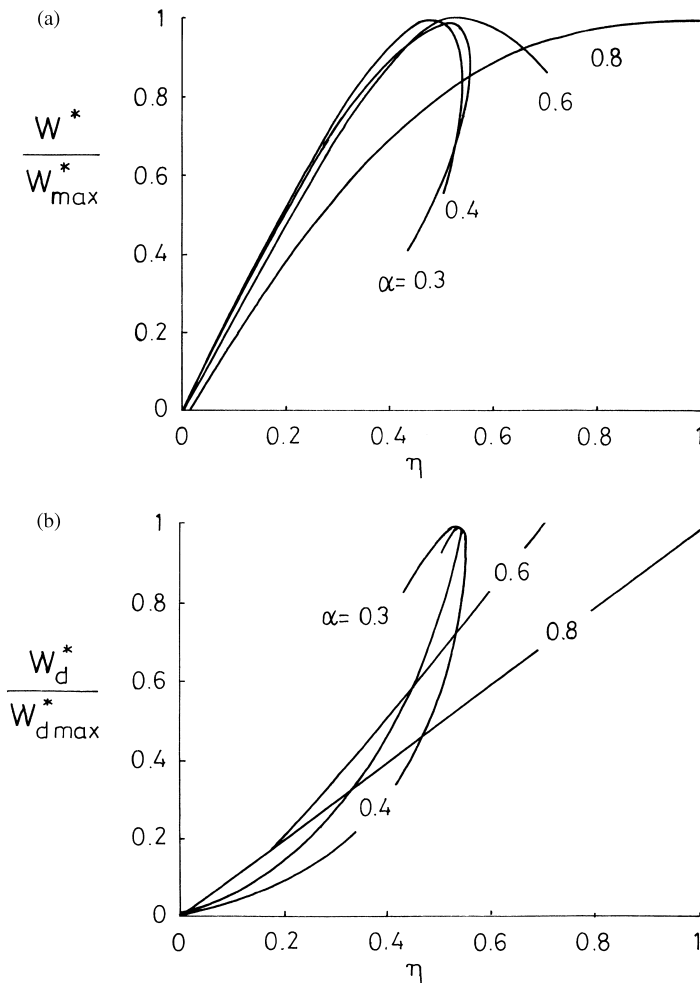


Fig. 4. (a) The normalized power and (b) normalized power density versus efficiency with respect to the cycle temperature ratio α ($\eta_t = \eta_c = 0.85$, $\eta_r = 0.80$, $r_t = 1.25$).

W_d^*/W_{dmax}^* values of the regenerative gas turbine cycle with the isothermal heat addition at the conditions considered by Ref. [13]. It is seen that the curves cannot easily be distinguished. Obviously, the problem noted by Ref. [13] seems to be eliminated by the isothermal heat addition to the regenerative gas turbine cycle considered in the present study.

Fig. 4 considers the effect of the temperature ratio. As it is read in Fig. 4(a) and (b), the efficiencies become 0.50 and 0.53 for $\alpha = 0.3$, 0.48 and 0.54 for $\alpha = 0.4$. The very interesting aspect of these working conditions arises from the comparison of the optimum cycle pressure ratios for each case. For the maximum power density conditions having the temperature ratio in the range 0.2–0.4, the optimum cycle pressure ratio $r_{p,mpd}$ varies only between 1.89 and 1.36 whereas $r_{p,mp}$ varies from

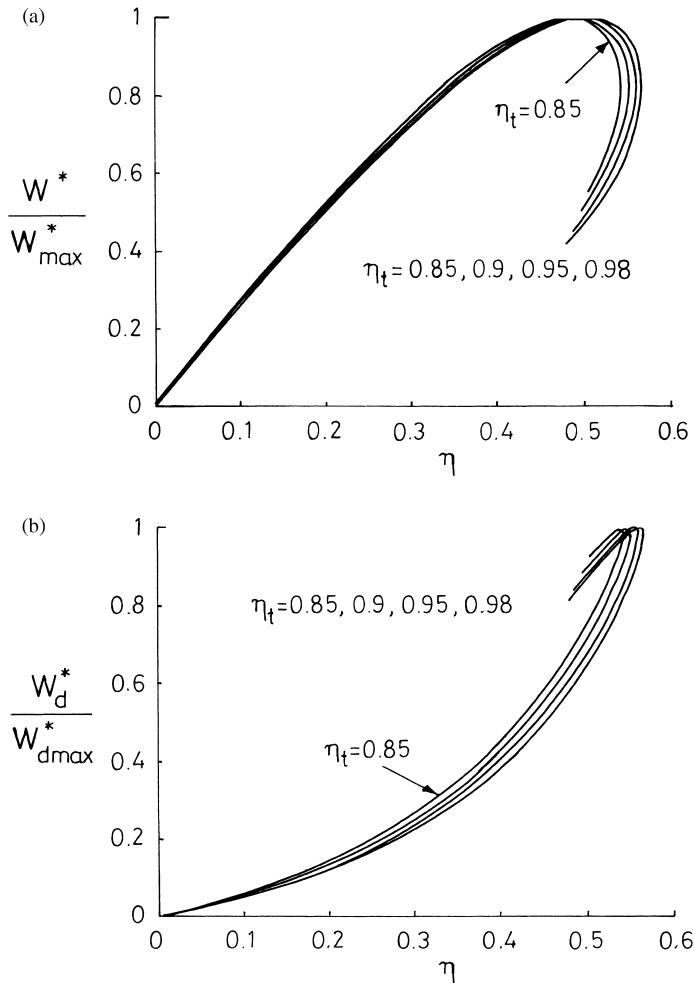


Fig. 5. The variation of the cycle efficiency η for various isentropic efficiencies of the turbine η_t at (a) normalized power and (b) normalized power density conditions ($\eta_c = 0.85$, $\eta_r = 0.80$, $\alpha = 0.4$, $r_t = 1.25$).

11.79 to 3.5 under the maximum power conditions. This means that the designs considering the maximum power density conditions have resulted in highly flexible designs against the changes of the cycle temperature ratio.

The values of the normalized power, $W^*/W_{\max}^*$ and normalized power density, $W_d^*/W_{d,\max}^*$ were calculated at different irreversibility levels of the turbine by considering various isentropic turbine efficiencies. The results are given in Fig. 5(a) and (b) as functions of the cycle efficiency. When the turbine efficiency decreases from 0.98 to 0.85, the cycle efficiency at the maximum power varies from 0.50 to 0.48 while the cycle efficiency at maximum power density varies from 0.55 to 0.53. Under these conditions, the optimum cycle pressure ratio $r_{p,\text{mp}}$ varies in the range 4.5–3.5 at

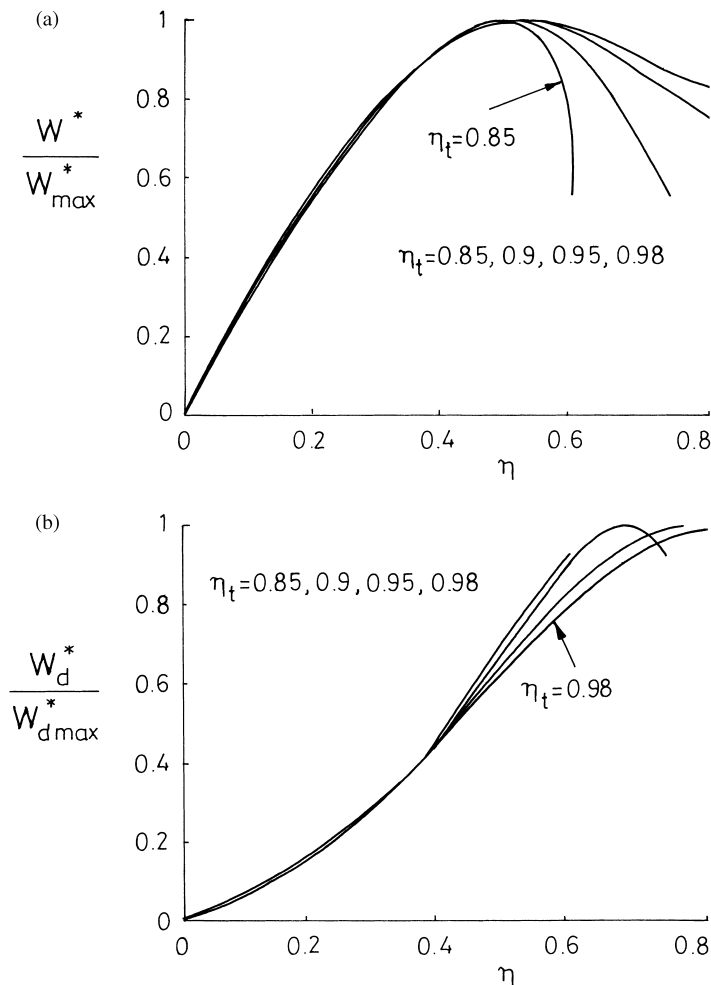


Fig. 6. The variation of the cycle efficiency η with respect to the regenerator efficiency η_r at (a) normalized power and (b) normalized power density conditions ($\eta_t = \eta_c = 0.85$, $\alpha = 0.4$, $r_t = 1.25$).

the maximum power conditions and $r_{p,mpd} = 1.60 - 1.36$ at the maximum power density conditions. It is seen that the efficiencies are higher whereas the optimum pressure ratios are smaller at the maximum power density conditions, than that for the maximum power conditions. The different isentropic efficiencies for the compressor were also considered and almost the same numerical results were obtained.

Fig. 6(a) and (b) summarize the effect of the regenerator efficiency η_r on the cycle efficiencies at the maximum power and the maximum power density conditions, respectively. The variations of the regenerator efficiency were considered in the range 0.98–0.85. The cycle pressure ratio has not changed and stayed at 3.5 at the maximum power. At the maximum power density, the cycle pressure ratio stayed at 1.36 and does not change, either. On the other hand, the efficiencies at the maximum power change from 0.53 to 0.49 while the values 0.85–0.60 are obtained at the maximum power density. For all cases, the regenerator efficiency has an important effect on the cycle efficiency, especially for the maximum power density conditions. It is noted by Medina et al. [13] that the efficiency at the maximum power density is greater or smaller than the efficiency at the maximum power depending on the value

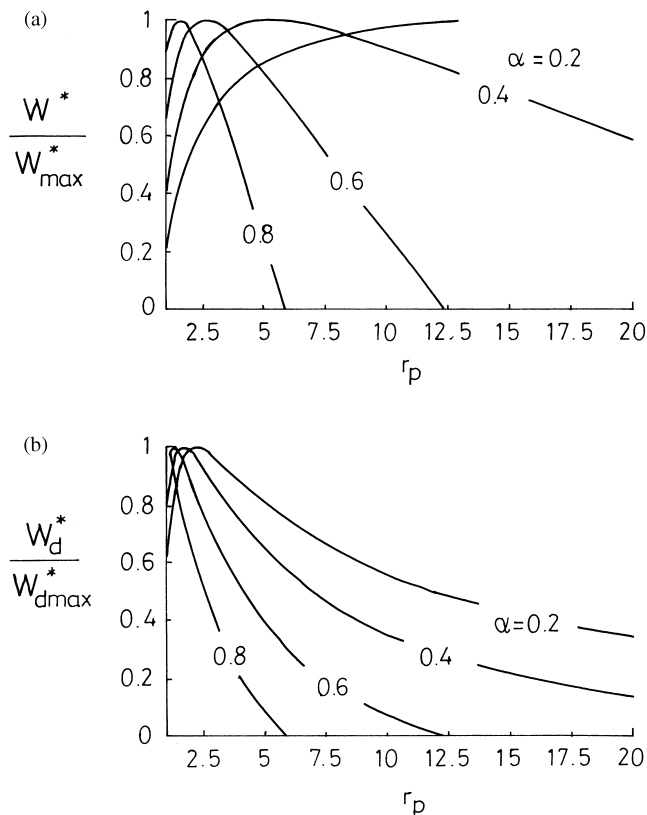


Fig. 7. The variations of (a) normalized power and (b) normalized power density according to the cycle pressure ratio with the consideration of the effect of temperature ratio α ($\eta_t = \eta_c = 0.95$, $r_t = 1.25$).

taken for the regenerator efficiency η_r . As can be seen in Fig. 6, the isothermal heat addition has really modified the Brayton cycle through the high efficiencies in all cases.

The combined effect of the cycle pressure ratio and the temperature ratio are investigated and the results are summarized in Fig. 7(a) and (b) by presenting the variations of the normalized power and normalized power density, respectively. Obviously, small values of the temperature ratio correspond to large temperature differences between the minimum and the maximum cycle temperatures. At a small temperature ratio, $\alpha = 0.2$, the power [Fig. 7(a)] increases, whereas at high temperature ratios, like $\alpha = 0.6$ or 0.8 , the power immediately decreases with increasing pressure ratio. The normalized power density has its highest values at low pressure values, as given in Fig. 7(b). For the temperature ratio of 0.2 , the maximum power density can be obtained at a pressure ratio almost five times smaller than the one for the maximum power condition and the power density decreases by increasing the cycle pressure ratio.

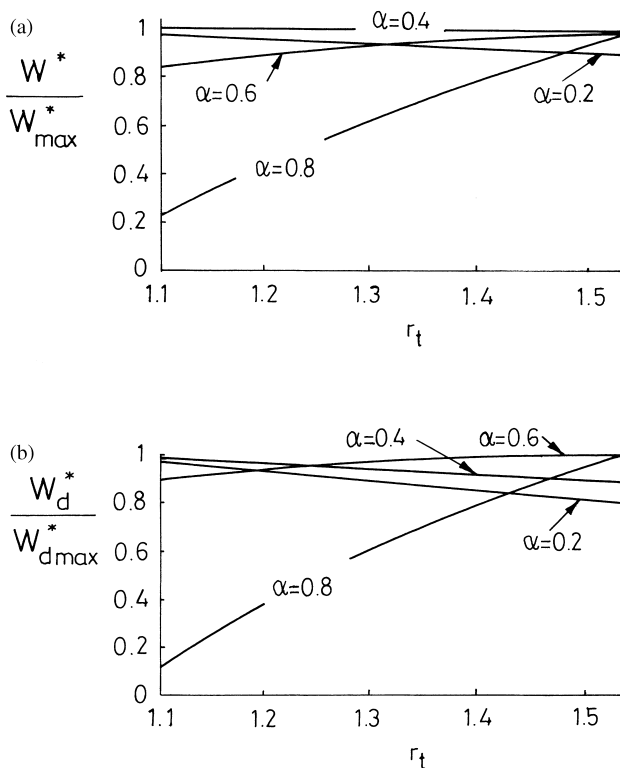


Fig. 8. The effect of isothermal pressure ratio r_t on the (a) normalized power and (b) normalized power density in accordance with the temperature ratio α ($\eta_c = 0.95, r_p = 5$).

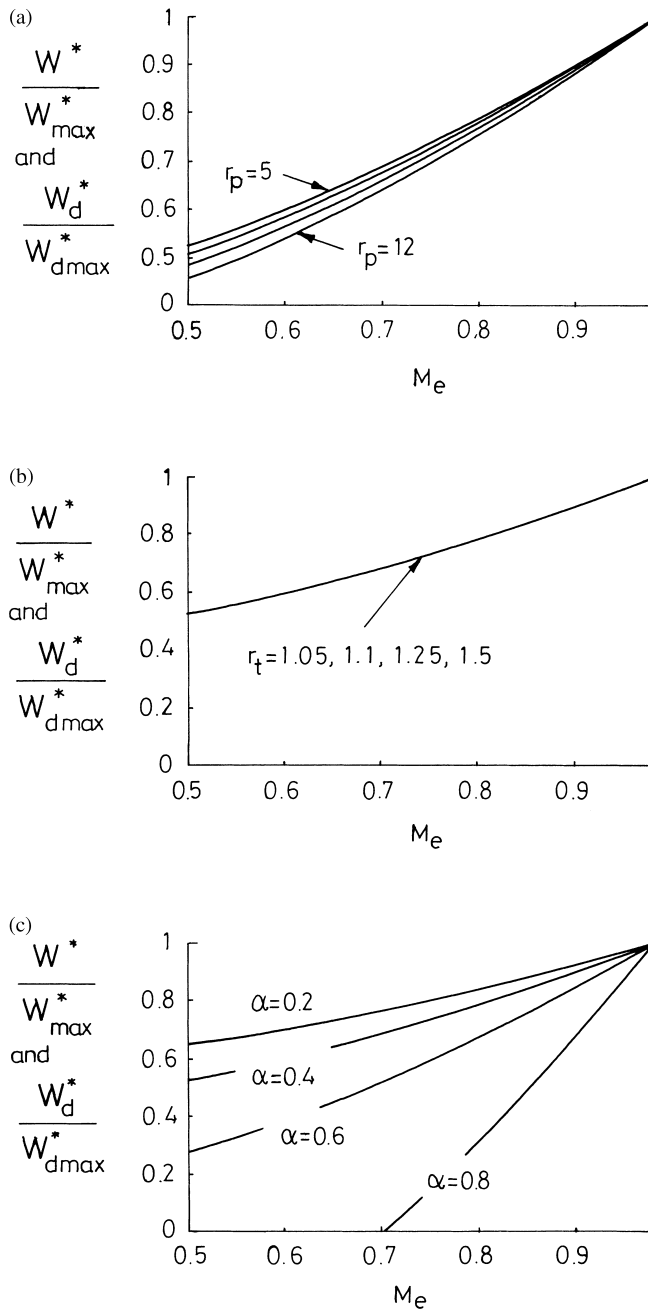


Fig. 9. The variations of normalized power and normalized power density according to the exit Mach number $> M_e$ with respect to (a) the cycle pressure ratio r_p ($\eta_t = \eta_c = 0.95, \alpha = 0.4, r_t = 1.25$), (b) the isothermal pressure ratio r_t ($\eta_t = \eta_c = 0.95, \alpha = 0.4, r_p = 5$) and (c) the temperature ratio α ($\eta_t = \eta_c = 0.95, r_p = 5, r_t = 1.25$).

The present study considers the isothermal heat addition, which is characterized by the isothermal pressure ratio, r_t . Fig. 8(a) and (b) summarizes the effect of the isothermal pressure ratio in accordance with the temperature ratio. Unlike the previous results, the power and power density have increased as the isothermal pressure ratio increases at high temperature ratios like $\alpha = 0.6 - 0.8$. A decrease is observed at small temperature ratios with the increase of the isothermal pressure ratio. The variation of the power and power density strongly imply that the isothermal heat addition process has a critical importance especially at small temperature differences between the minimum and the maximum cycle temperatures, i.e. at high α values. As it is seen from the curve for $\alpha = 0.8$, a little increase in r_t results in considerable increase in power level.

The Mach number at the exit of the converging combustion chamber (M_e) is an indicator of the power generation rate at the turbine. Due to the changes in the flow conditions, M_e changes. In Fig. 9(a),(b) and (c), the effects of variations of the exit Mach number on the normalized power and power density by considering the cycle and the isothermal pressure ratios and the temperature ratio have been summarized, respectively. Obviously, it is seen that the exit Mach number has the same effect on both of the normalized power and power density and causes the power levels to increase in the same manner; that is a single curve is identified. The normalized power and power density increase with increasing exit Mach number. Fig. 9(a) shows that the decrease in the pressure ratio makes little increase in the power levels. But the effect of the changes in the isothermal pressure ratio on the power and power density cannot be identified, as seen in Fig. 9(b). At constant cycle and isothermal pressure ratios, the cycle temperature ratio yielded a definite effect on the power and power density. The high values of the power and power density have been obtained for the temperature ratio $\alpha = 0.2$ and continued to increase with increasing M_e [Fig. 9(c)]. When the curve for $\alpha = 0.8$ is considered, it is found that the Mach number at the converging combustion chamber does not have a value smaller than 0.7.

4. Conclusions

The purpose of this study has been to design a more efficient small-sized gas turbine engine and to determine the optimal operating conditions. A regenerative gas turbine engine with isothermal heat addition, working under the frame of the Brayton cycle, has been considered. The optimization has been carried out by using the maximum power density method and an irreversible turbine, compressor and regenerator have been considered. The design parameters of the engine under the maximum power and the maximum power density conditions have been derived numerically.

Obviously, the low efficiency problem noted for the regenerative Brayton cycles under the maximum power density conditions has been eliminated by the isothermal heat addition to the regenerative gas turbine cycle. More tolerable designs against the changes of the cycle temperature ratio and the inefficiencies of the cycle

components have been obtained. Finally, the results have shown that the regenerative gas turbine engine with isothermal heat addition designed according to the maximum power density condition gave the best performance and exhibited the highest cycle efficiencies.

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